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SUMMARY

Curves of forced frequency against amplitude are presented for the conditions where the forced frequency is both increased and decreased into the resonant range. On the basis of these curves it is shown that the practical resonance frequency is the point where wrinkling first occurs and that the resonance frequency will be subject to considerable travel once permanent wrinkles appear in the vibrating shell. The decreasing mode of striking resonance is found to be by far the most destructive condition.

INTRODUCTION

The problem of airplane vibrations, always a sore point in the design of aircraft, has of late become of a new and greater importance. Although this change in importance has been due in large part to the pronounced increase in flying speeds, the problems have certainly not been simplified by the trend to all-metal forms of construction. The component parts of an airplane are now generally heavier than before and consequently have lower natural frequencies; in addition, the new constructions have lower inherent damping ability than did the old fabric structures.

Strangely enough, these new conditions have been greeted, not by investigation of the vibration properties of the new structures, but by redoubled efforts toward keeping out of the resonant conditions. This method is unquestionably the best means now known of combating the vi-

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bration problem, but it was felt that a knowledge of the vibration characteristics of shell structures is a first requisite to the successful execution of the technique. Accordingly, the present work was done in order to fill at least partially the need.

The experiments were conducted in the Aeronautics Laboratory of the University of Michigan, under the supervision of Mr. Burdell L. Springer. It was decided to investigate monocoque cylinders in torsion mainly because the work would have direct application to the case of a fuselage vibrating torsionally, a condition often met in practice. It will be seen at once, however, that the results can be applied to any steel structure inasmuch as all fail through elastic instability.

THEORY AND PRESENT PRACTICE

The monocoque cylinders upon which these tests were made were restrained against any bending and thus represent spring members having but a single degree of freedom. Elementary vibration theory easily treats of such a case and the results have been well verified by experiment, provided that the deflection of the spring member be always proportional to the load. In this case the amplitude (or deflection) simply builds up as resonance is approached until the failure deflection is reached and the spring member breaks or deforms. Typical curves for such a system in torsion are reproduced in figure 1. The curves are plotted for a constant damping coefficient, each curve referring to a different magnitude of exciting torque. Theoretically, failures will occur whenever the amplitude reaches the static failure deflection, a value denoted by the horizontal line A. It should make no difference whether resonance is approached from the low- or the high-frequency side. If the system is undamped, each curve will have a vertical asymptote at $\frac{\omega}{p} = 1$. Since the curves of figure 1 are for a damped system, each curve will reach some maximum that will depend upon the amount of damping.

It is on the assumption that this theory will hold for elastically unstable structures that present practice regarding airplane vibrations is based. Since it is practically impossible to compute the natural frequency of a part such as a wing or fuselage in the various modes, the airplane must first be completed and then tested for frequencies. This procedure is usually carried out by means of a small portable outfit consisting of an eccentric weight rotated by a variable-speed motor of some sort through a flexible drive. The eccentric weight assembly is so clamped to the airplane as to vibrate the part investigated; power is then supplied throughout a large range of frequencies until the resonance hump is obtained, the resulting curve being one such as B, figure 1. The probable frequencies to be encountered are known from experience so that it can be told at once whether the part is safe or must be redesigned. Since a necessarily small power is used to operate the vibrator, the structure is never elastically unstable during the test.

Regarding the vibration of members that do not deflect in a linear proportion to the load, such theory could hardly be expected to hold. If, for instance, the load-deflection curve should be one such as figure 2, and if the amplitude should extend above that corresponding to the break in the curve, some changes in the vibration curves are almost certain to occur. A thin-wall monocoque cylinder in torsion - or for that matter, any thin-wall monocoque structure - gives just such a curve, the break in it being due to the development of waves or wrinkles. Should the shell in question have relatively thick walls (such as would be the case in a full monocoque airplane fuselage), the first break in the load-deflection curve would, in all probability, represent the formation of a permanent wrinkle or kink. The subsequent behavior of the shell under vibration should here also be out of the scope of elementary theory, as will later be shown.

TEST APPARATUS AND METHODS

The test work of this investigation was done by means of a vibrating machine especially constructed for the purpose. Details are shown in figure 3 and photographs in figure 4. The cylinder to be tested is clamped to two end plugs, which are provided with sockets and rivets spaced around the circumference so as to insure good retention of the specimen. One plug is rigidly secured to a concrete pillar, the other is bolted to a vertical arm. A steel shaft, held in concrete pillars at either end, supports the arm-and-plug assembly and insures that no bending will occur under vibration. The vertical arm is supported on this same shaft at two points separated by some 6 inches so that longitudinal rocking of the arm is prevented. Hence the arm is allowed freedom to rock in only the direction that will produce torsion of the cylinder. An eccentric weight in the form of an adjustable rod, rotated by an upper shaft, generates the oscillations. They are transmitted to the arm through a self-alining bearing set in it and holding the upper shaft. Power is supplied by a 220-volt motor of 3-horsepower capacity.

Since the frequency of forced vibration was the speed of the motor, it was measured by means of a tachometer mounted coaxially and in back of the motor shaft. Amplitudes were large enough to be read visually with fair degree of accuracy. The indicator for this purpose was rigidly attached to the top of the vertical arm, and the scale was fixed to the adjacent pillar. Readings were taken from a distance of about 6 feet in order to lessen the parallax effects. All the component parts of the apparatus were examined for their own natural frequencies to insure that these frequencies did not fall in the operating range.

The cylinders tested were all of the same size, being of 10-inch diameter and having a total length of 12 inches, of which two 1-inch sections were taken up by the end plugs. The material was commercial 0.010-inch duralumin (17ST), the true thickness of which was determined by micrometer as 0.011 inch. Test specimens were constructed so that rolling marks were at right angles to the polar axle. The joint consisted of a single row of aluminum rivets spaced 1 inch apart.

Preparatory to each test, the eccentricity of the rod was adjusted to a very small value and a test made to determine the natural frequency of the cylinder acting as an elastically stable body. The subsequent test procedure consisted of simply varying the speed of the motor by means of a field rheostat and of taking simultaneous revolution speed and amplitude readings, the weight eccentricity being adjusted to suit for each test.

A series of tests was run starting from a low revolution speed and working up into resonance. Because of a peculiarity of behavior discovered during this first series, a few additional tests were made in which the speed was decreased into the resonance range from a high value, such tests being made at first on cylinders which had already failed, and the results finally checked by a similar operation on a sound one.

RESULTS

The first series of tests run was for the purpose of mapping out resonance curves of increasing frequency. In these tests the power supplied was gradually increased from zero, which meant that, in general, the frequency of forced vibration also gradually increased. The curves of figure 5 are the result. Each curve is for a different weight eccentricity or degree of unbalance of the weight as expressed in inch-ounces. The lowest of the three curves shows a behavior such as might be expected from any spring not subject to instability. Here no noticeable wrinkling or failure occurred because the weight eccentricity was quite small. The retention of high amplitude at frequencies past resonance was due to the use of constant weight eccentricity throughout the curve; that is, the applied torque varied with the square of the revolution speed. Because all interesting phenomena occurred within a small range of speed, this shortcoming of the apparatus was not found to be of significance.

The other two curves represent cases where the cylinders not only wrinkled under vibration but also suffered a loss in torsional rigidity in consequence of the amplitudes imposed upon them. This latter effect was considered as constituting a failure. Upon examination of the specimens, evidence of such failure was found to exist in the form of very minute kinks in the surface of the shell. These were so slight as to escape notice under a cursory visual inspection.

Because of the peculiar behavior of the cylinders in these two cases the exact time of formation of the permanent wrinkles is not known. In both tests power was gradually increased until point M was reached. Here the rapid development of a very loud rattle announced the wrinkling of the shell. The segment from M to N was the shell's own doing, no power adjustments being possible because the time consumed in the process was of the order of about one-tenth of a second. Naturally, the shape of this segment was indeterminate as far as concerns the present apparatus, and the line shown is merely a reasonable guess. Since the curve from M to N goes in a direction of decreasing frequency, it was apparent that the permanent kinks in the cylinder walls were formed somewhere along this segment. On the basis of the crescendo nature of the sound produced it is believed that only wrinkling within the elastic limit occurred in the region around point M, but that this soon developed into the more pronounced condition wherein permanent kinks are formed. An equilibrium condition was reached at N and no further failure occurred, a fact borne out by the decrease in amplitude with increasing forced frequency.

From these two curves it will be seen at once that resonance for a cylinder under torsional vibration is largely a function of the magnitude of the torque acting. The smaller this torque becomes, the closer to the elastic resonance point will failure occur and the smaller will the loop become. In all cases, however, the real resonance point will occur somewhat to the left of the point where $\frac{\omega}{p} = 1$ because the effect of damping on low torque curves is to move the maximums to the left.

During the course of the test work of curve D (fig. 5), a reverse run was taken after failure. Starting from the right end, point O, at a motor speed of 1,200 r.p.m., the power was gradually decreased. Curve D was found to be faithfully retraced until point N was reached. Here the usual loud rattle seemed to indicate a condition of resonance. However, further decrease in power from this point, far from reducing the amplitude as might be expected, caused it to rise continually with falling speed - the result being to divest the cylinder of practically all the strength that it might still have had. A practical limit to this building up of amplitude was encountered at around 500 r.p.m., where the available power of the motor dropped below the power required to continue the large amplitudes. Consequently, the pronounced vibration ceased abruptly when this condition was reached.

The same procedure was repeated on subsequent failed cylinders of the series always with the same result. Finally, a reverse run was made with a perfect cylinder. In this case the vibrating arm was rigidly held in order to prevent failure of the shell during the time required for the motor to reach 1,200 r.p.m.

The results of all the reverse runs are presented in figures 6 to 9 with dotted curves of the ascending tests added for the purpose of comparison. That the decreasing mode of striking resonance is by far the most destructive can be seen at once from an examination of these figures. It is also clear that from the time large amplitudes are obtained down to the low-speed drop-off point, the cylinder is always in or near a resonant condition. It will shortly be shown that during these descending tests the cylinder cannot be in resonance but must of necessity be at some point of slightly higher frequency.

Consider figure 8, the reverse run made with a perfect cylinder. Power was gradually decreased until at around 900 r.p.m., the rapid development of noise indicated first the elastic wrinkling of the shell, and subsequently, the formation of the permanent kinks which represent an initial degree of failure. Unlike the ascending tests, no automatic shifting of frequency on the part of the cylinder occurred; the system was quite obviously at equilibrium at this speed and this particular amplitude. Now if this condition were exactly that of resonance, further decrease of power must inevitably cause reduction of amplitude. Because resonance was certainly not encountered above 900 r.p.m. and because further drop in motor speed, even with attendant decrease in power, served to increase the amplitude, it follows that the 900 r.p.m. point must be above the resonance condition of the cylinder in the particular degree of failure obtained there. The subsequent increase in amplitude as the descent progressed served continually to deepen the wrinkles in the shell, so that the natural frequency of the system always stayed below motor speed. Consequently, when the power finally became too small to continue the large amplitudes, the drop-off occurred and resonance never was actually reached, even though an amplitude of almost 0.01 radian was obtained.

In view of this traveling tendency of the resonance point as previously described, the explanation of the loop in the ascending speed curves is now clear. If reference is again made to curve D (fig. 5), it will be seen that the segment from the initial failure point M to the stable point N is nothing more than a modified descent curve. At or slightly above M, large amplitudes, with the accompanying wrinkles, are encountered. The intermittent formation of the wrinkles introduces a new damping component, which seems to be rather high. As the power supplied is constant, this new damping action serves to decrease the power available to produce rotation of the eccentric weight and the reverse curve is followed as a consequence. Point N is an equilibrium condition where the damping and vibration power is equal to the power amount supplied. Increase in power from point N goes into building up the rate of revolution of the weight and causes the curve to proceed to the right once more, thus completing the loop.

The mode of failure of the cylinders is perhaps deserving of some mention. Cylinders that failed under an amplitude of 0.009 radian, while known from static tests to possess permanent wrinkles, seemed perfectly sound when examined visually. So slight was the extent of wave formation that it could only be detected by the sense of touch and even then with difficulty. The failure pictures presented (fig. 10) are of cylinders that were subjected to amplitudes of about 0.02 radian solely for photographic purposes. In all cases the waves were exactly parallel to the polar axis of the cylinder. At the end of each wave a small S-shaped kink was formed, which was found by observation to be necessary for the formation of cross-directional waves produced as the cylinder twisted from one side to the other.

CONCLUSIONS AND GENERAL REMARKS

From what has gone before it is now possible to make a few observations concerning the behavior of monocoque cylinders under torsional vibrations.

1. The "elastic" resonance point as measured in the customary way is of no significance if the vibration power to be encountered is high enough to produce permanent wrinkles, however slight. The resonance point in such an instance will fall considerably as soon as wrinkling occurs and the effects upon the cylinder or fuselage will be ruinous.

2. Insofar as the possibility of failure is concerned, the resonance frequency should be considered as the frequency where wrinkling first occurs; this frequency will depend upon the magnitude of the alternating forces acting. (See fig. 5.)

3. Because of the traveling tendency of the resonance point, the case where wrinkling occurs while the forced frequency is falling, is by far the most destructive condition. In practice this condition would occur if a low-

wing monoplane were gradually to decelerate. A tail buffeting would here occur in which the frequency of forced vibration would be dropping with the speed.

4. Although the results of this investigation apply rigorously to shells only, they may be applied to some extent to semimonocoque structures if the skin of such structures is designed to carry a large percentage of the total load.

5. The power supplied by the motor for these tests varied in some direct relation to the load. In practice, the power of forced vibration is likely to be just as great in the low range as in the high. Hence the effects discussed will probably be even more pronounced than the tests show.

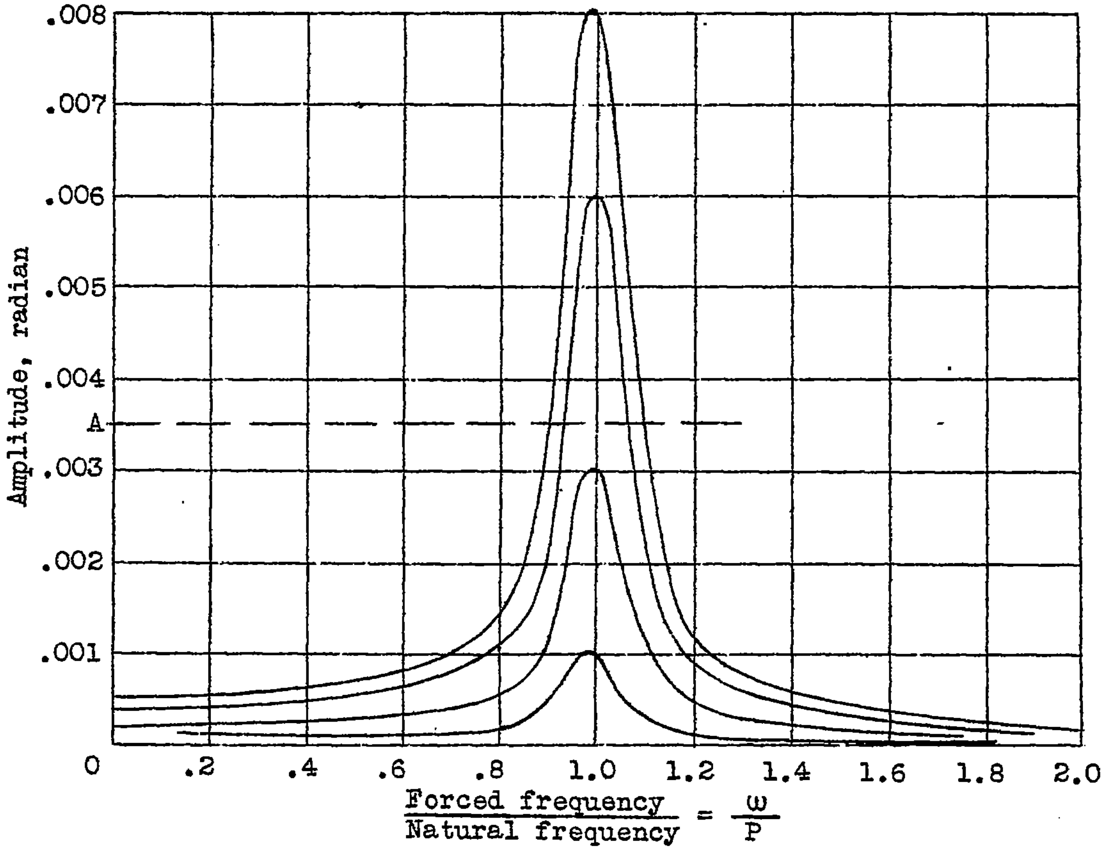


Figure 1.- Typical elastic resonance curves.

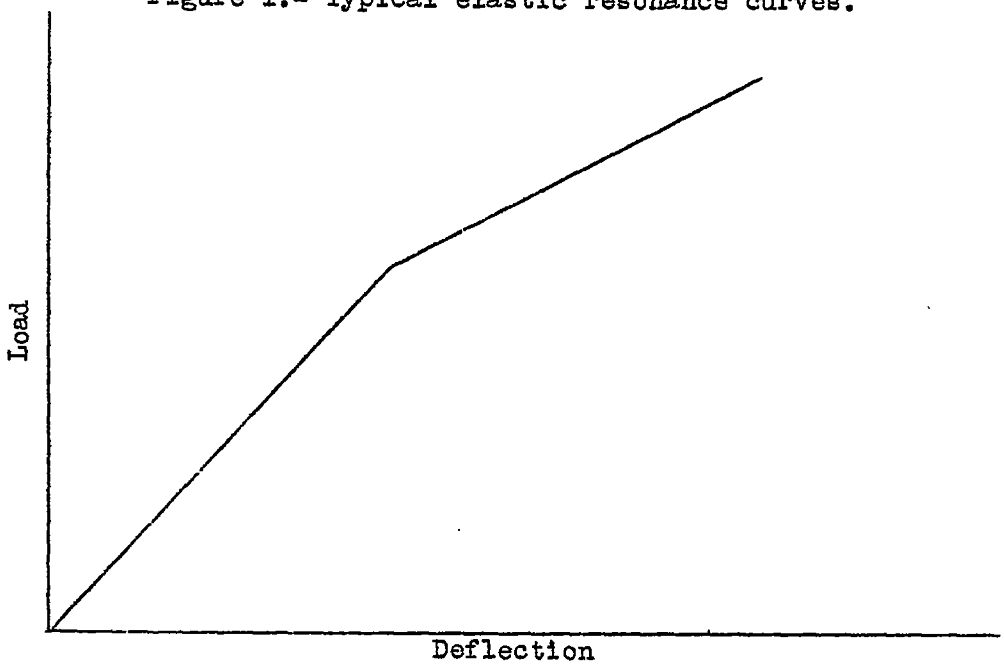


Figure 2.- Typical load-deflection curve of a shell.

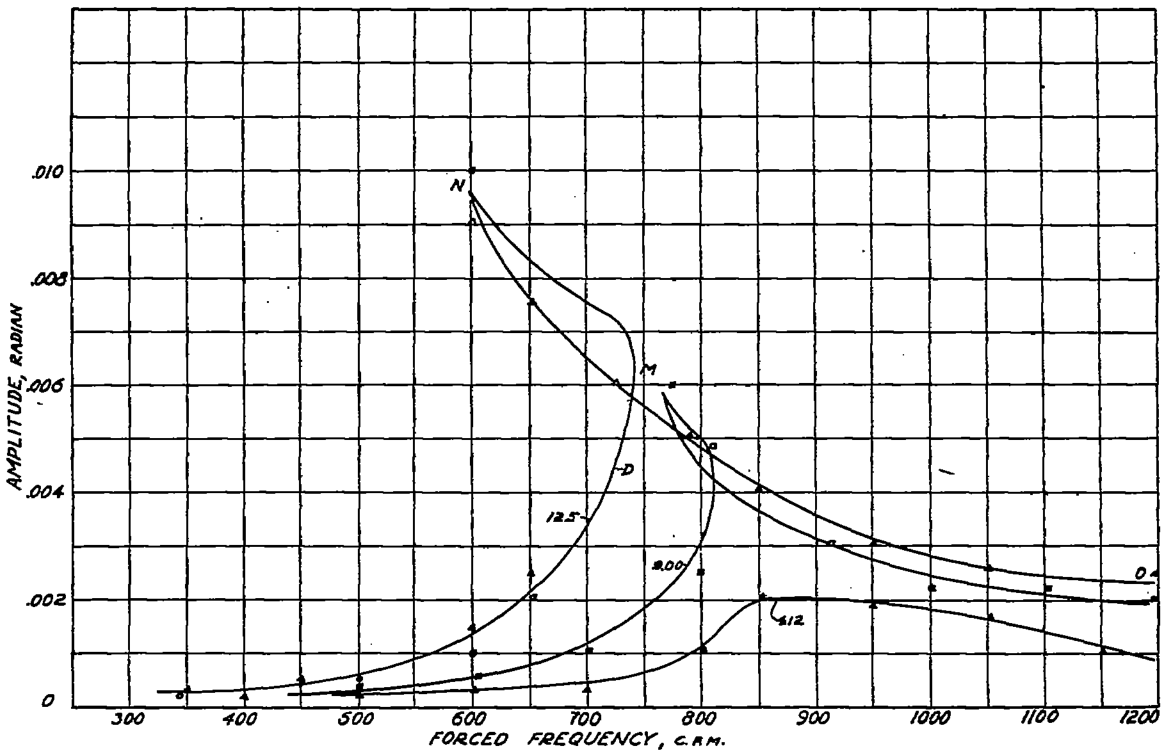


FIGURE 5. ASCENDING CURVES FOR WEIGHT ECCENTRICITIES OF 4.12, 9.00, + 12.5 IN. OZ.

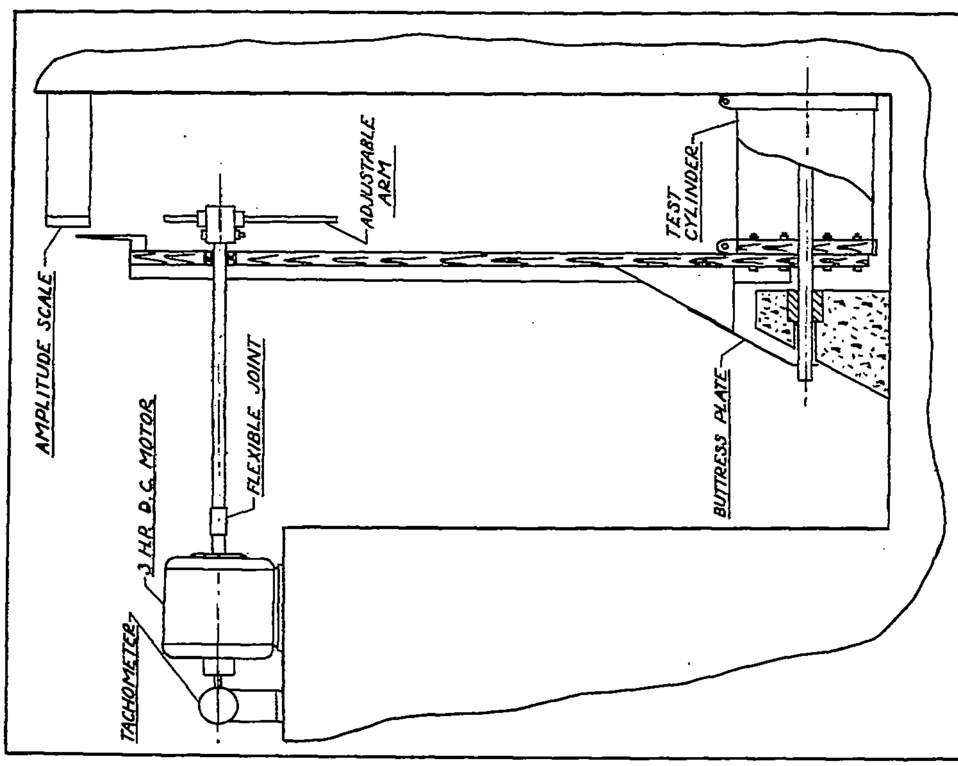
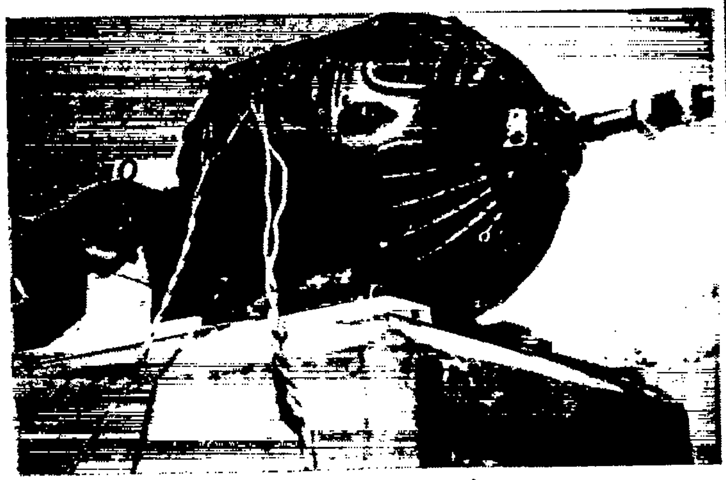
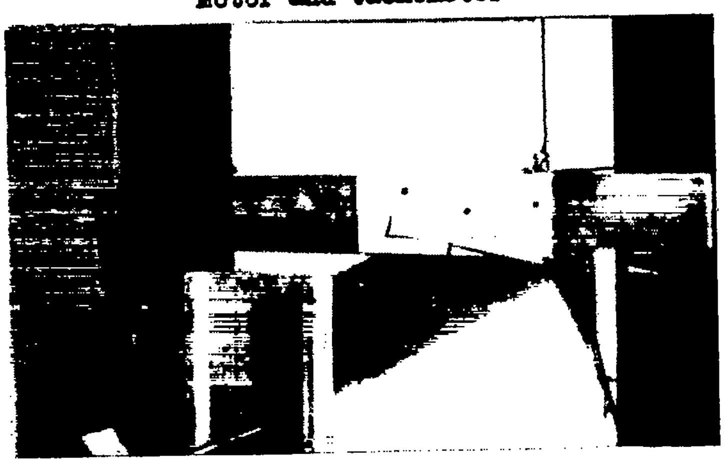


FIGURE 3.- TEST APPARATUS.
NOMINAL CYLINDER LENGTH - 10 IN.
DIAMETER - 10 IN.
MATERIAL - 0.011 IN. DURALUMIN
SCALE - 1½" = 1'

General layout



Motor and tachometer



Amplitude scale

Figure 4.- Detail photographs of test machine

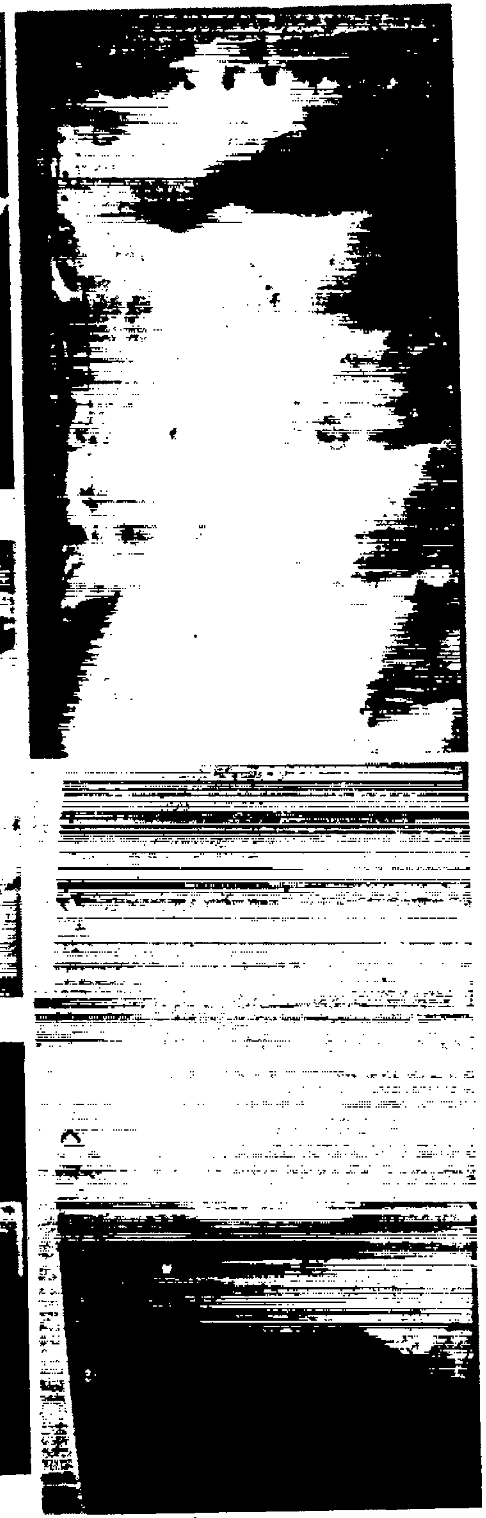


Figure 10.- Typical failures (Cylinders unrolled).

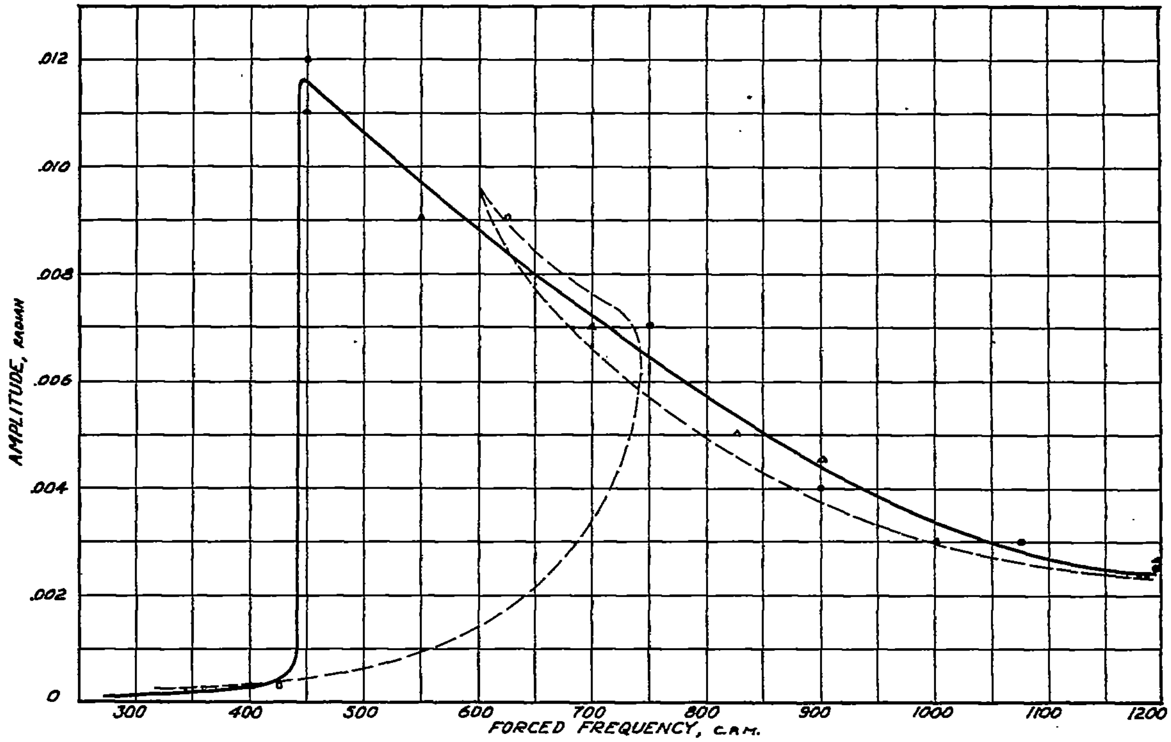


FIGURE 6 - DESCENDING TEST OF FAILED CYLINDER, W. E. = 12.5.

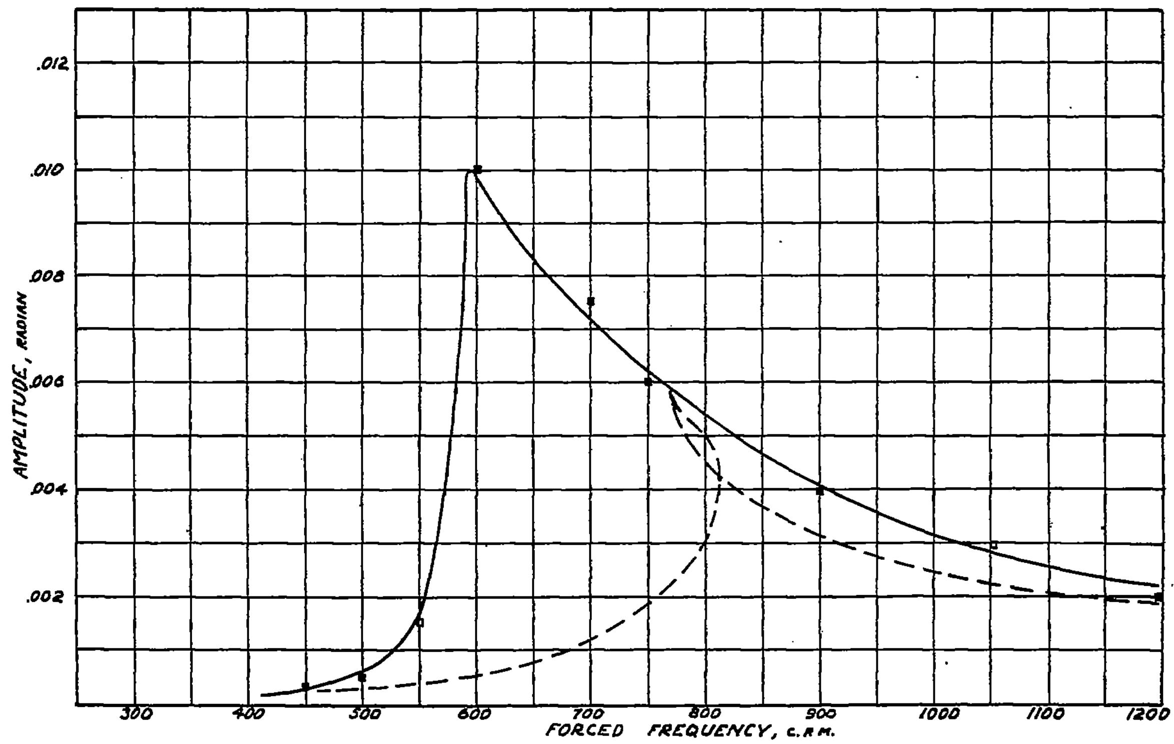


FIGURE 7. DESCENDING CURVE OF FAILED CYLINDER, W. E. = 9.00 INCH OUNCES

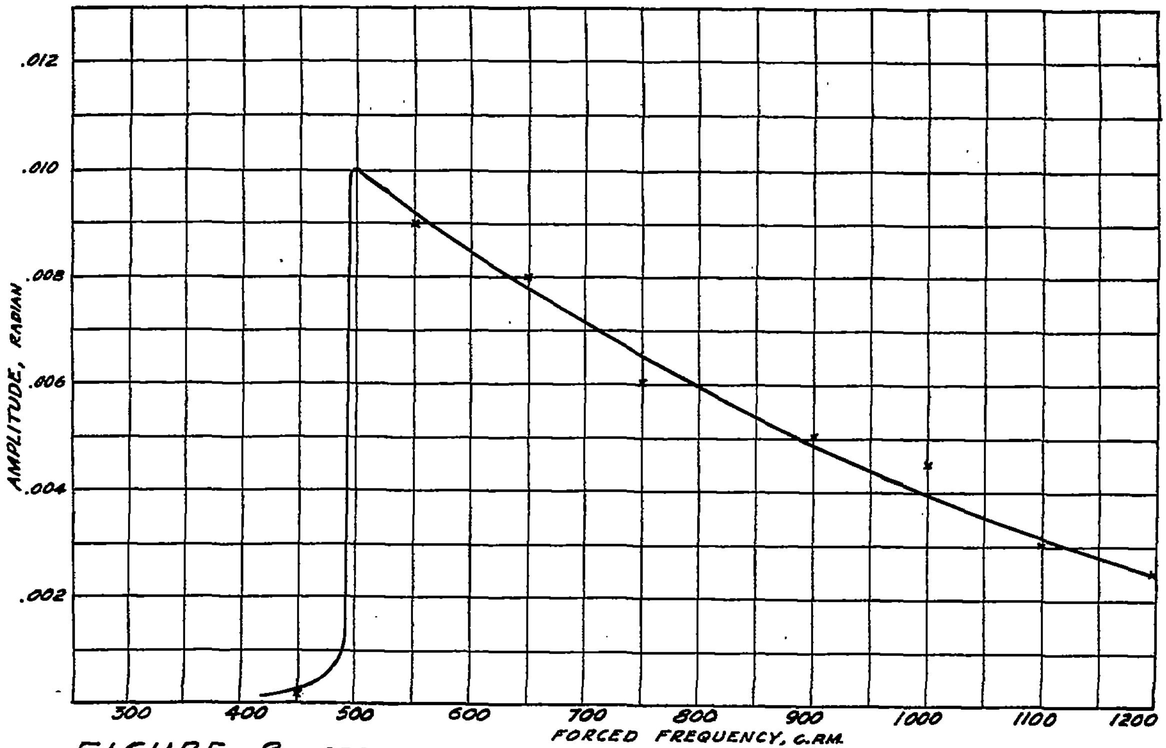


FIGURE 8 DESCENDING CURVE FOR SOUND CYLINDER, W.E. = 9.00 INCH-OUNCES

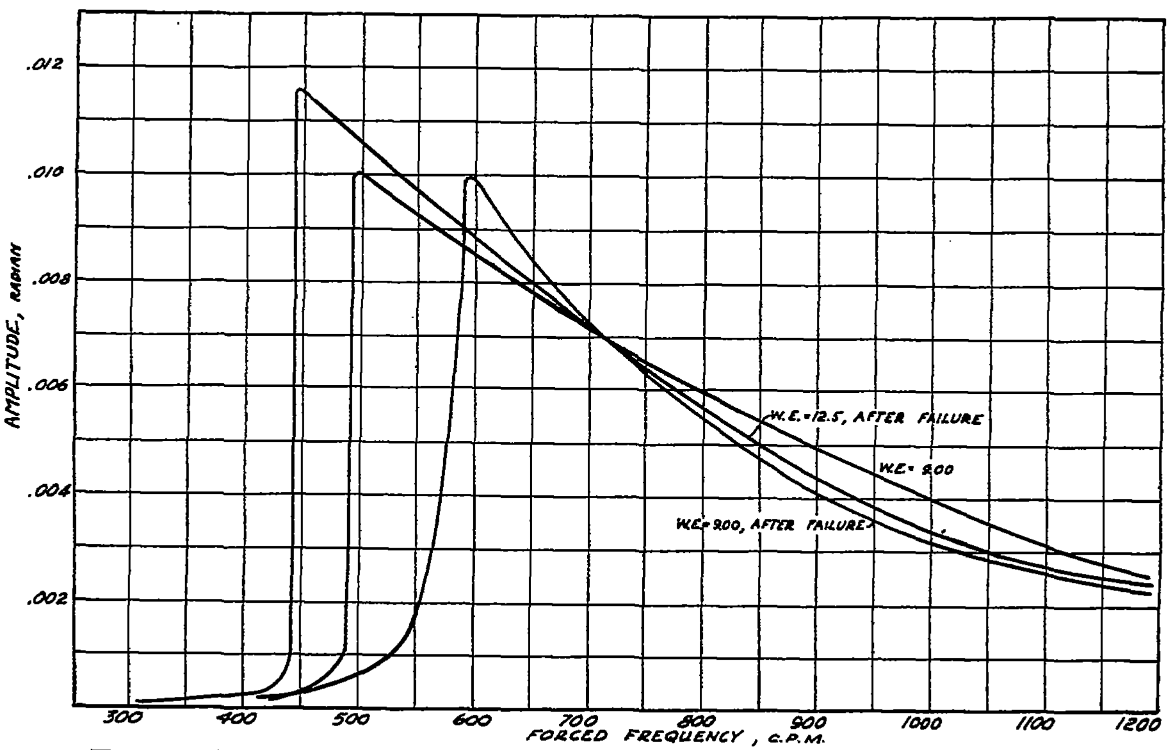


FIGURE 9 SUMMARY OF DESCENDING CURVES